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Melting ice with air bubblers

Kevin L. Carey

Introduction

The author, a civil engineer, is a member of CRREL's Construction Engineering Research Branch.

Air bubblers have been used to melt floating ice for a long time, the first installation having been made over 65 years ago. While many installations have been very successful, the systems have been designed largely on a hit-or-miss basis. This is because until recently there had been no detailed engineering analysis of air bubbler operation or the conditions under which a bubbler succeeds or fails. In the last ten years, however, interest in winter navigation has resulted in much research into ways of suppressing ice. We now have a clearer understanding of air bubbler systems and how they can operate successfully. What follows relies largely on a report by Ashton (1974), which should be consulted, along with two others (1978 and 1979), for a much more thorough discussion.

Air bubbler systems are used to melt ice in harbors, to keep structures like piers and docks free of ice, to minimize ice formation in critical bends in navigation channels, and to make the movement and storage of ice easier during icebreaking operations. Bubblers transfer heat energy stored in the water up to the underside of the ice cover. The heat may melt the ice

1. Typical bubbler system arrangement.
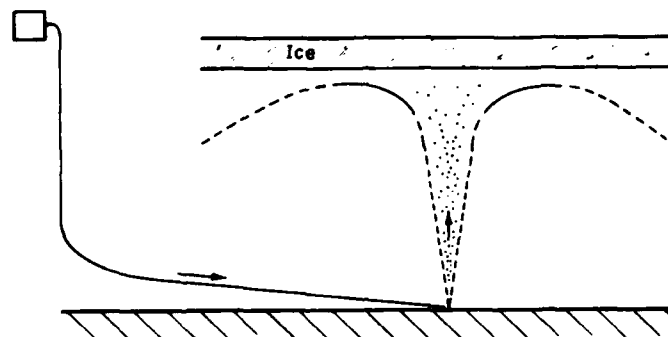
completely, creating an open-water area above the bubbler, or it can reduce its thickness, so that ships can pass through it or so the ice breaks up earlier and more easily.

Bubblers transfer heat to the ice cover through the circulation of water caused by a rising plume of air bubbles released from orifices well beneath the surface (Fig. 1). The water from the lower levels, usually warmer than that near the ice cover, brings heat to the ice and causes melting. Under favorable conditions, the energy going into melting can be many times greater than the energy used to operate the system. It is this multiplication effect that makes air bubblers so advantageous over any direct use of energy to melt ice.

However, an air bubbler system can operate successfully only when the characteristics of the site are favorable. Several factors have to be known before any bubbler installation can be planned.

Site conditions

It should be obvious that the physical dimensions of the water body need to be known. A fairly good idea of the depth and bottom topography (plus the type of bottom materials, for anchoring purposes) is necessary. The depth and the horizontal dimensions, when combined with the temperature of the water, give an estimate of the heat that can be drawn on to melt the ice.

Water temperature is of prime importance. It can't just be assumed that the temperature at the bottom is 4 °C (39 °F), the maximum-density temperature. While theoretically this would be expected, actual temperature measurements of water bodies in winter (which are unfortunately very few) have shown that in most water bodies where bubblers might be installed, actual bottom water temperatures are much less, as low as 0.1° to 0.2 °C (32.2° to 32.4 °F) or even 0.01 °C

(32.02 °F). Field investigations indicate that only in very deep lakes or very small, sheltered lakes do bottom temperatures approach 4 °C (39 °F). And in a water body with any significant amount of flow or circulation, such as a river or a bay open to a larger water body, the temperature may be at 0 °C (32 °F) throughout the depth as a result of turbulent mixing, although temperatures as high as 0.5° to 1.0 °C (32.9° to 33.8 °F) have been found by Wortley in some bays of the Great Lakes (G. Ashton, personal communication, 1982).

Natural movement of the water, through either flow or circulation, is important to evaluate, not only because of the chance of having no available reserve of heat, but also because the flow will affect the water circulation created by the bubbler. The expected location of melting can be shifted horizontally, with the result that the actual amount of warmer water moved up to the ice cover becomes diluted and less effective. Naturally, flow is expected in rivers. But flow may be present in lakes or harbors too, due to the inflow of tributary streams, wind-induced circulation, or currents from a large water body connected to a bay or harbor.

To evaluate the feasibility of a bubbler system, some knowledge of the natural ice cover characteristics is needed. For example, the normal ice thickness should be known, so that it can be predicted whether the bubbler-induced melting will create open water or simply thin the ice to some equilibrium thickness. In the case of open water, the expected size of the opening is related to the normal ice thickness. Also, it is critically important to know if the ice cover will be stationary or moving. If it is moving, the melting will be spread over a large area of ice but there may be so slight a reduction in ice thickness as to be of no benefit. Expected dates of ice cover formation and break-up should also be determined.

Finally, the information about site conditions should include climatic data. The air temperature at the site will influence the heat transfer through the ice cover. The amount of snow on the ice cover also governs the heat transfer rate, because its insulating effects will keep the top surface of the ice warmer than the air if the air temperature is below 0 °C (32 °F). The best indicator of temperatures affecting an ice cover is the freezing index, the total number of degree-days of freezing. The more detailed the freezing index information the better, e.g. average cumulative totals of degree-days during the winter, or consecutive daily records. Useful additional

climatic information includes wind data, cloud cover data, precipitation records, etc., although such data are not readily factored into the currently available mathematical analysis of air bubbler operation.

Bubbler installation and operation

Mechanical Details. The size and type of air compressor used in a bubbler installation depend on many factors. The size and depth of the bubbler will dictate the air requirements. Whether the installation is permanent or not, and where it is located, will help to determine the type of compressor used. Compressors commonly fall in the range of 0.0094 m^3/s (20 ft^3/min) to 0.71 m^3/s (1500 ft^3/min) in delivery rates, at pressures up to 3.4×10^5 Pa (50 psig). It is sometimes determined that an air dryer is needed in the system, to prevent ice formation at the orifices. In order for this decision to be made, the expected relative humidity of the air at the site, plus the pressure drop at the orifices, will have to be considered.

Piping from the compressor is made up of the supply line and the diffuser line. Standard design approaches govern the choice of the supply line, balancing economy and performance. Naturally, it is helpful to keep the supply line short to minimize frictional losses, but the most convenient location for the compressor station will probably be the main factor setting the supply line's length. So its diameter should be chosen to permit pressures high enough at the end of the supply line to ensure air discharge at the most distant orifice of the diffuser line. A rapid-emptying fitting is often supplied at the far end of the diffuser line, to permit rapid removal of water at start-up.

The diffuser line may present a design challenge in some installations. There is a step-wise variation in air flow and velocity as each orifice is passed, and there is a gradual variation in pressure along the length of the line. As suggested above, air must reach the orifice farthest downstream, and pressure has to be higher than the water pressure at the level of the line. In addition, a fairly constant air discharge rate per unit length of line is desired from one end to the other. For installations that are large or that have nonuniform orifice spacing or diffuser pipe sizes, a detailed analysis of the diffuser is needed (see Vigander and others, 1970). But for most installations of modest size, the variations in air discharge along the pipe are negligible. Berlamont and others (1973) and Camp

and others (1968) give techniques that evaluate variations with fixed orifice spacing and pipe size.

As a general rule, orifice spacing is chosen to be half the depth of submergence of the diffuser. This spacing gives reasonably good results approximating a line-source of air, as opposed to separated point-sources. Orifice size depends on the desired air discharge rate, and is determined by the conventional orifice discharge equation:

$$sQ_a = Q_o = C_d \frac{\pi d^2}{4} \sqrt{2\Delta p / \varrho_a}$$

where s = orifice spacing
$\quad Q_a$ = air discharge rate per unit diffuser length
$\quad Q_o$ = air discharge rate from one orifice
$\quad C_d$ = orifice discharge coefficient
$\quad d$ = orifice diameter
$\quad \Delta p$ = difference between air pressure within the diffuser and water pressure outside the diffuser
$\quad \varrho_a$ = mass density of the air within the diffuser.

Plume Mechanics and Melting. Considering the bubbler as a line source of air, the plume it creates is two-dimensional. Figure 2 shows the velocity distribution of the plume (a Gaussian distribution), which increases linearly with height above the vicinity of the diffuser line. Because of flow conditions near the orifices and the initial momentum of the air jet, the
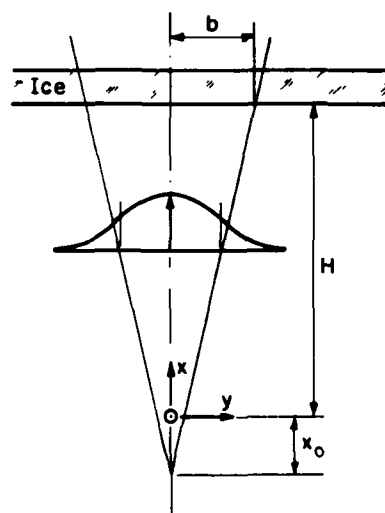
2. Definition sketch for a two-dimensional bubbler plume.

linear boundaries of the plume have a theoretical vertex about 0.8 m (2.6 ft) below the diffuser (denoted by x_o in Fig. 2). The expansion of the plume results from drawing in steadily increasing amounts of surrounding water (entrainment), all driven by the momentum of the buoyant rising bubbles. It is this entrainment that delivers heat from the presumably warmer water at depth to the underside of the ice, and causes melting. The water flow in the plume per unit length (Q_w) at the surface can be described in terms of Q_a, air discharge per unit length, and H, the depth of the diffuser:

$$Q_w = 5.53 Q_a^{0.5}(H+0.8)^{0.5}\left[\log_e\left(1 + \frac{H}{10.3}\right)\right]^{0.5}$$

which can be approximated by:

$$Q_w = 2.28 Q_a^{0.5} H^{0.813}.$$

This is in meter-second (SI) units, and is for sea level, though the effect of other elevations is minor.

As shown in Figure 2, the plume is $2b$ wide when it reaches the surface. The measure of b is chosen as the standard deviation of the Gaussian velocity distribution of the plume at the surface. So the width of the plume in meters can be found by:

$$2b = 0.364(H+0.8)Q_a^{0.15}.$$

After the plume strikes the underside of the ice, or reaches an open water surface, the flow spreads horizontally outward from the area of impingement. It is not important for this discussion to examine this horizontal flow, other than to say that its velocity drops off very rapidly, and the principal melting effects of the bubbler occur in and near the impingement region of width $2b$ as defined above.

What is important is what happens to the horizontal flow after that, i.e. the induced circulation within the water body. For the bubbler to continue to melt ice over a long period, it must draw on the thermal reserve not just in the vicinity of the bubbler, but over a broad area. Its ability to do this depends on the circulation induced in the water body, a subject that is not well understood.

Two possibilities have been envisioned, a "filling" scheme and an "overturning" scheme. In the filling model, the pre-

sumably less dense water resulting from bubbler operation accumulates at the surface, gradually displacing the denser fluid downward. In the overturning model, the plume sets up circulation cells, with the plume's flow plunging back to the level of the diffuser line at some distance to the side of the bubbler. Work by Baines and Turner (1969) indicates that circulation takes place in water bodies whose depth is greater than about one-fourth their width, and that the filling model is more likely to be followed in shallower or broader water bodies.

Recent studies (G. Ashton, personal communication, 1982) suggest that circulation and complete mixing take place within a horizontal distance of 5 to 7 times the depth for any bubbler depth, regardless of the width and depth characteristics of the water body. Outside this circulation zone, it appears that exchange of water still occurs via layered flow toward and away from the circulation zone. Thus a combination filling and overturning scheme forms a third possible model. In a strict filling model, the thermal reserve would be used by the bubbler at an initial high rate which decreases with time as the thermal reserve is used up. In a strict overturning model the circulation cells would prevent heat from being drawn into the bubbler from beyond the first cell. Thus the actual available thermal reserve would be much less and would be depleted much earlier. In a combination model, the thermal reserve consumption would lie between these two extremes.

How much heat is delivered to the surface by the bubbler? This of course depends on the water flow delivered to the surface, Q_w, and on the temperature profile in the water from the surface down to the depth of the diffuser. Although not strictly accurate, it is an acceptable approximation to take the temperature of the plume as it reaches the area of impingement as the depth-weighted average of the temperature profile between the surface and the diffuser line, T_{avg}. Then the heat delivered to the surface is

$$G = Q_w(T_{avg} - T_s)\varrho_w c_p$$

where G = heat delivered per unit time per unit length of the two-dimensional bubbler

Q_w = plume flow at the surface (described earlier)

T_s = temperature at the surface (0 °C in the case of an ice cover)
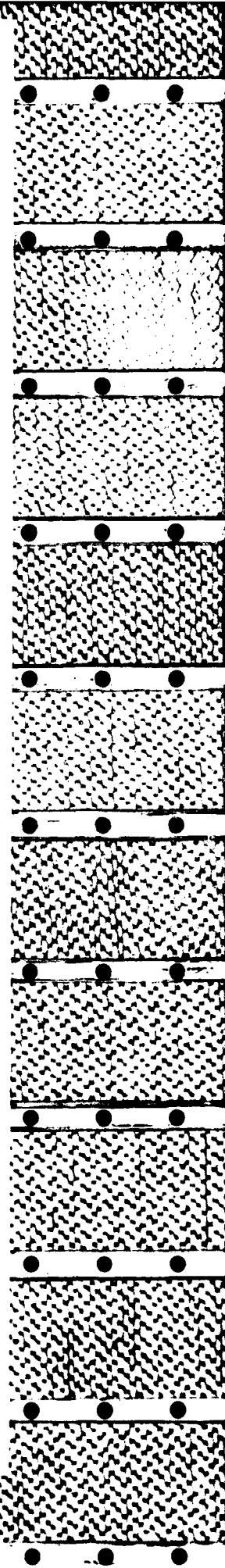
ϱ_w = mass density of water

c_p = water's specific heat capacity.

It must be remembered that the temperature profile, and hence T_{avg}, change with time as the bubbler continues to operate.

As most of the heat G remains with the water in circulation, we need to know how much is actually transferred to the ice cover. By using an analogous analysis for a slot jet of air impinging on a plate, Ashton developed relationships to yield the heat transfer coefficient within the impingement region. In terms of Q_a and H, the heat transfer coefficient in W m^{-2} °C^{-1} (SI units) is

$$h_b = 17,220[\log_e(H+10.3) - 2.332]^{0.31}(H+0.8)^{-0.69}Q_a^{0.16}$$

which can be approximated by

$$h_b = 6888H^{-0.384}Q_a^{0.16}.$$

This coefficient is considered constant over the width $2b$, and drops off very rapidly to either side of the bubbler. Thus the actual heat transferred to the ice cover (in W m^{-2}) in the impingement region is

$$q_b = h_b(T_{avg} - T_s)$$

or

$$q_b = h_b \cdot T_{avg}$$

since $T_s = 0\,°C$ for an ice cover.

What does this mean in terms of thinning an ice cover or maintaining an ice cover at an equilibrium thickness? The answer lies in the basic heat balance equation at the ice/water interface,

$$q_b - q_i = \varrho_i \lambda (d\eta/dt)$$

where q_i = heat conduction upward through the ice (and through a snow cover if present) to the atmosphere

ϱ_i = mass density of the ice

λ = latent heat of melting

$d\eta/dt$ = melting rate of the ice of thickness η.

If q_b is greater than q_i, then the ice cover will become thinner, while if $q_b = q_i$, it will maintain an equilibrium thickness.

The calculation of q_i is straightforward, and won't be discussed here. It should be noted, however, that a snow cover will reduce q_i significantly, and that snow cover thicknesses and the conductivities of ice and snow will need to be known.

Two examples may serve to illustrate the application of the foregoing information. First, suppose a bubbler is capable of transferring q_b = 200 W m^{-2} to the ice cover, and that the mean daily air temperature (an acceptable substitute for the temperature of the top surface of a snow-free ice cover) is $-25\,°C$. The equilibrium ice thickness η_e would be found by the relation

$$q_b = \frac{k_i(T_s - T_m)}{\eta_e}$$

where k_i = 2.24 W m^{-1} deg^{-1}, the conductivity of pure ice; T_s = $-25\,°C$, the top ice surface temperature; and $T_m = 0\,°C$, the bottom ice surface temperature. Then η_e is found to be 0.28 m. (If instead q_b was 400 W m^{-2} or 100 W m^{-2}, then η_e would be 0.14 m or 0.56 m, respectively.)

The second example considers a thinning ice cover. Suppose Q_a = 0.001 m^2 s^{-1} and H = 10 m, such that h_b = 979 J m^{-2} s^{-1} deg^{-1}, and T_{avg} = 0.1$\,°C$. Then q_b = 97.9 W m^{-2}. Now, for simplicity, assume that the top surface ice temperature is 0$\,°C$, so that q_i = 0 W m^{-2}, and the entire amount of q_b goes to melting. Then $q_b = \varrho_i\lambda d\eta/dt$. Taking ϱ_i = 916 kg m^{-3} and λ = 3.34$\times$10^5 J kg^{-1}, then $d\eta/dt$ becomes 3.12$\times$10^{-7} m s^{-1} or 2.76 cm per day of melting, a bit more than an inch a day.

Rationale of Operation. Any prospective bubbler installation should first be subject to a calculation of the maximum theoretical thermal reserve available for melting ice. Then this should be translated into the maximum theoretical volume of ice melting. What this will show, in many cases, is that the amount of ice suppression that can be achieved is not as extensive as expected, or in other words, that the thermal reserve is a scarcer commodity than first thought. This scarce thermal reserve must be used cautiously and prudently.
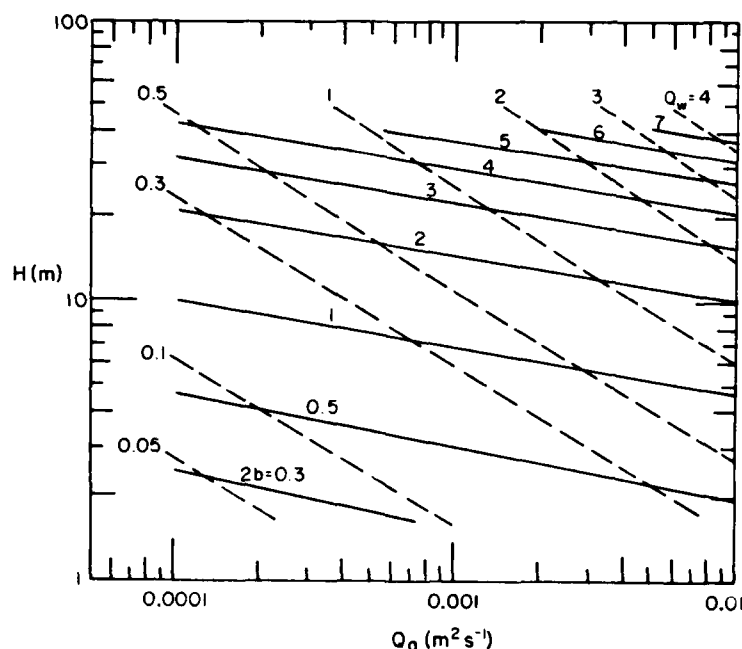
Two operating schemes can be chosen, singly or combined, to wisely exploit the reserve. First, instead of seeking the creation of open water, most system objectives can be met just as well by achieving a thinned ice cover. This greatly reduces heat wasted to the atmosphere, and helps to save the thermal reserve so that it can be drawn upon over a longer period of time. Similarly, intermittent operation of the bubbler system, rather than continuous operation, will extend the usefulness of the thermal reserve. Qualitatively matching periods of bubbler operation to times when air temperatures produce the greatest potential ice growth would be particularly beneficial, but even an arbitrary on–off schedule is helpful.

Performance expectations

The performance of a bubbler system is dependent on the air flow rate Q_a and the depth of submergence H, both of which the designer can choose, plus the water temperature and thermal reserve, which he can't. To illustrate the expected values of some of the important parameters, Table 1 gives values of Q_w, the water flow per unit length reaching the impingement region, in $m^2 s^{-1}$ ($ft^2 min^{-1}$); $2b$, the width of the impingement region, in m (ft); and h_b, the heat transfer coefficient, in $W m^{-2} °C^{-1}$ (Btu $ft^{-2} s^{-1} °F^{-1}$); all for three typical values of air flow Q_a and five typical values of depth of submergence H. The same information (except heat transfer coefficient) is portrayed in Figure 3.

Table 1. Bubbler performance parameters for three air flow rates (Q_a) and five submergence depths (H).

H		Q_w		2b		h_b	
(m)	(ft)	($m^2 s^{-1}$)	($ft^2 min^{-1}$)	(m)	(ft)	($W m^{-2} °C^{-1}$)	(Btu $ft^{-1} s^{-1} °F^{-1}$)
Q_a = 0.01 $m^3 s^{-1}$ (6.5 $ft^3 min^{-1}$)							
2	7	0.39	250	0.51	1.7	2.37×10^1	3.76×10^{-1}
6	20	0.98	630	1.2	3.9	1.73×10^1	2.74×10^{-1}
10	33	1.5	970	2.0	6.6	1.42×10^1	2.25×10^{-1}
20	66	2.6	1700	3.8	12	1.04×10^1	1.65×10^{-1}
40	130	4.4	2800	7.4	24	7.37×10^2	1.17×10^{-1}
Q_a = 0.001 $m^3 s^{-1}$ (0.65 $ft^3 min^{-1}$)							
2	7	0.12	78	0.36	1.2	1.64×10^1	2.60×10^{-1}
6	20	0.31	200	0.88	2.9	1.20×10^1	1.90×10^{-1}
10	33	0.47	300	1.4	4.6	9.80×10^2	1.55×10^{-1}
20	66	0.83	540	2.7	8.9	7.19×10^2	1.14×10^{-1}
40	130	1.4	900	5.3	17	5.10×10^2	8.08×10^{-2}
Q_a = 0.0001 $m^3 s^{-1}$ (0.065 $ft^3 min^{-1}$)							
2	7	0.039	25	0.26	0.85	1.14×10^1	1.81×10^{-1}
6	20	0.098	63	0.62	2.0	8.27×10^2	1.31×10^{-1}
10	33	0.15	97	0.99	3.2	6.78×10^2	1.07×10^{-1}
20	66	0.26	170	1.9	6.2	4.97×10^2	7.88×10^{-2}
40	130	0.44	280	3.7	12	3.53×10^2	5.60×10^{-2}

3. Water flow (Q_w, $m^2 s^{-1}$, dashed lines) and impingement width (2b, m, solid lines) as functions of air discharge and bubbler depth.

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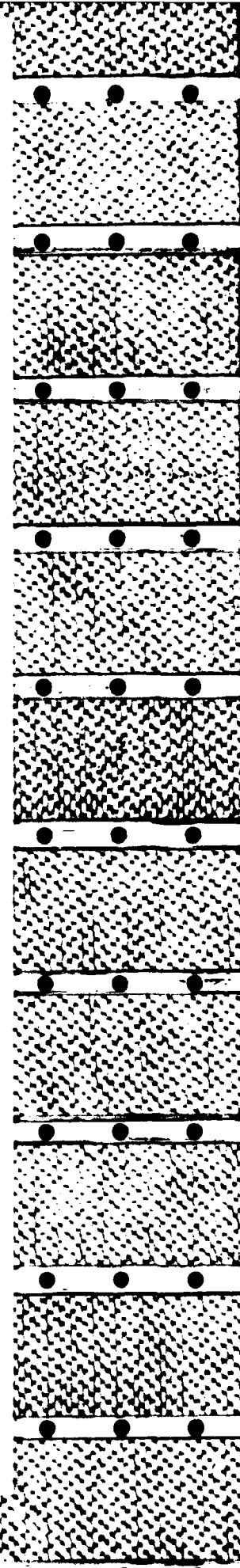
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